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Title	Model scale testing of the Tupperwave device with comparison to a conventional OWC
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Publication date	2018-09-25
Original citation	Benreguig, P., Murphy, J. and Sheng, W. (2018) 'Model scale testing of the Tupperwave device with comparison to a conventional OWC', Proceedings of ASME 2018 37th International Conference on Ocean, Offshore and Arctic Engineering, Madrid, Spain, June 17–22, Volume 10: Ocean Renewable Energy, OMAE2018-78611 (10pp). doi: 10.1115/OMAE2018-78611
Type of publication	Conference item
Link to publisher's version	https://asmedigitalcollection.asme.org/OMAE/proceedings/OMAE2018/51319/V010T09A048/278143 https://archive.asme.org/events/omae2018 http://dx.doi.org/10.1115/OMAE2018-78611 Access to the full text of the published version may require a subscription.
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OMAE2018-78611

MODEL SCALE TESTING OF THE TUPPERWAVE DEVICE WITH COMPARISON TO A CONVENTIONAL OWC

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ABSTRACT

A model testing campaign of the Tupperwave device was carried out to prove the working principal and validate a numerical modelling that had previously been developed. An appropriate and challenging scaling method was applied to the floating device to correctly model the air compressibility in the Tupperwave PTO. In parallel, a model scale conventional OWC was also built using the same axisymmetric structure geometry and both devices were tested and compared. The testing showed that the Tupperwave device produced less average useful pneumatic power than the conventional OWC. The primary losses were attributed to pneumatic power dissipation through the valves. The pneumatic power delivered by the Tupperwave device was however significantly smoother.

The paper describes the experimental set-up and the methods used to assess the devices performance. The results provide a direct comparison between the two physical models pneumatic power performances and an in-depth analysis of the valves behaviour is shown.

INTRODUCTION

Wave energy has the potential of playing an important role in the world renewable energy mix [1]. A large variety of wave energy converters (WEC) exist but a technology to efficiently and economically harness wave energy has not emerged yet.

Among this variety of WECs, the Oscillating Water Column (OWC) is one of the most studied and shows interesting characteristics. The OWC device comprises of a partly submerged structure open at its bottom, inside which air is trapped above the water free surface. The hydraulic energy from the waves is first transferred to the air contained in the OWC chamber as pneumatic energy and then converted into mechanical and electrical energy by an air turbine and a generator. The success of the OWC largely depends on the performance and reliability of the air turbine.

Conventionally, the turbine connects the OWC chamber to the atmosphere. An alternating air flow is created through the turbine resulting from the rising and falling of the water column. Various types of pneumatic PTO systems can be used to harness this alternating air flow [2]. In order to deal with the inwards and outwards flows, the majority of OWC prototypes are equipped with self-rectifying bidirectional turbines, such as the Wells or impulse turbines. This type of turbine has a lower efficiency than a conventional unidirectional turbine. Also, as the airflow changes direction every couple of a seconds, the pneumatic power fluctuates.

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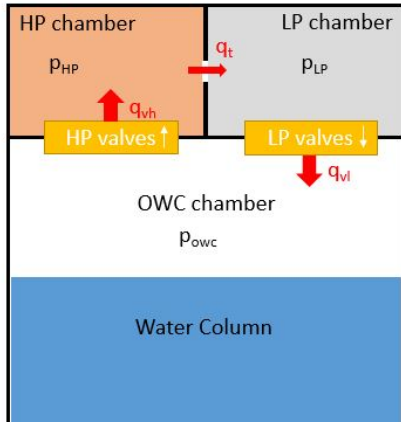


FIGURE 1. Schematic diagram of the Tupperwave device concept

tuations through the turbine are important and induce difficulties in terms of control and power quality. Moreover, as the turbine is opened to the atmosphere, significant acoustic noise can be created.

An alternative consists of creating a closed-circuit air flow in the device driven by the rising and falling of the water column and using rectifying non-return valves. The air flow is then harnessed with a high-efficiency unidirectional turbine. This closed-circuit method has been tested in different multi-chambers OWC devices, as is the case of the LEANCON, [3], the SeaBreath [4], or the ShoreSWEC [5]. Those very large multi-chamber devices work on the assumption that similar quantities of air are being blown from a chamber and sucked into another chamber at the same time.

The goal of the Tupperwave project is to design and validate at laboratory scale an innovative OWC concept that uses air compressibility in two large fixed volume accumulator chambers to generate a smooth unidirectional air flow in a closed circuit. Each of these chambers is connected with the OWC chamber through a system of non-return valves, and connected between them by a high efficiency unidirectional turbine. Figure 1 describes schematically the working principle of the device under study. The motion of the water column alternatively pushes air into the high pressure chamber (HP chamber) through the HP valves when rising, and sucks air out from the low pressure chamber (LP chamber) through LP valves when falling. The air flows rather steadily from the HP chamber to the LP chamber across an unidirectional turbine.

For the development of such a system, the concept was applied in a floating axisymmetric structure, with submerged dimensions based on a device previously tested in the MaREI Center as part of the Marinet project [6]. The PTO characteristics, mainly the accumulator chamber volumes and the turbine damp-

ing coefficient, were previously investigated in [7] to maximise the average pneumatic power whilst minimizing their fluctuations. It showed that the Tupperwave device could potentially create as much useful pneumatic power as a conventional OWC of the same geometry.

As a second step of the project, a tank testing campaign of the device at model scale was carried out in the Lir-National Ocean Test Facility's Deep Ocean Basin. The device was built at a scale of 1/24 and equipped with all necessary instrumentation to fully monitor its behaviour. A model scale conventional OWC was also tested using the same axisymmetric structure geometry. The two devices were tested in the same conditions both in fixed and floating states in regular and irregular sea states. The Tupperwave working principle relies on compressibility which is not scalable with Froude similarity law. Another scaling method for the HP and LP chambers was used. The multiple objectives of the physical testing were:

1. to prove the device working principal and assess the device power production
2. to show the feasibility of small scale testing of such floating device despite complex compressibility effects
3. to compare the Tupperwave and conventional spar OWC model pneumatic power performances
4. to validate the numerical models developed in [7].

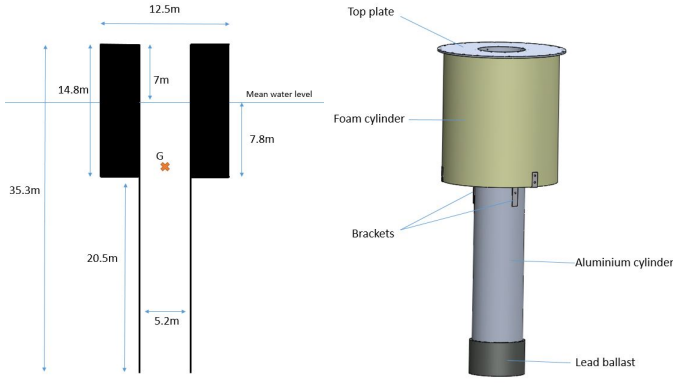
This paper presents the design, fabrication and tests of the Tupperwave model scale device and compares the performance with the tested conventional OWC. The validation of the numerical model is not in the scope of this article.

PHYSICAL MODEL DESIGN AND FABRICATION

In order to fairly compare the Tupperwave device to the conventional OWC, a single Spar buoy was built and both PTOs were built to fit on this same Spar. The following sections describe the Spar and PTO design and fabrication for both devices.

Floating Spar

The spar buoy concept is the simplest concept for floating OWC. It is an axisymmetric device consisting basically of a submerged vertical tail tube open at both ends, fixed to a floater that moves essentially in heave. The single heave motion makes it simple to model: The system can be represented as a two body system (structure + water column) moving relatively in the vertical direction. The geometry of the spar buoy at full scale is shown in fig. 2a. For the model scale, all dimensions were multiplied by a scaling factor $\epsilon = 0.0415$ which is close to 1/24th scale. The Spar model was built using a aluminium cylinder of 21.59cm internal diameter and 6.35mm thick. A 6mm aluminium plate was welded on top to support the different PTOs. The floatation consisted of high density foam. A 14kg lead ring was attached at the



(a) Full Scale Dimensions (b) Model Scale 3D design

FIGURE 2. SCHEMATIC OF THE SPAR BUOY

bottom of the float to ballast the device. The 3D-design of the spar is shown in fig. 2b.

Conventional OWC Power-Take-Off

The PTO of the conventional Spar OWC was chosen to be a self-rectifying impulse turbine. This type of turbine is commonly modelled physically by an orifice plate. Assuming constant air density under the testing conditions, the pressure Δp_t and air mass flow rate \dot{m}_t across the thin orifice plate are related by the following equation:

$$\Delta p_t = k_t \dot{m}_t^2 \quad (1)$$

k_t is called orifice damping coefficient. The numerical modelling carried out previously in [7] showed that the damping coefficient of $25 \text{ Pa.s}^2.\text{kg}^{-2}$ was close to optimal for the full scale device. For the model device, the damping coefficient was also scaled down using Froude scaling and 3 different orifice plates with damping coefficients close to this value were built and tested. Their exact damping coefficient were determined experimentally prior to testing in the OWC by forcing a known sinusoidal air flow across the orifice and measuring the pressure drop.

The orifice plate was simply screwed on top of the column using a rubber joint between the orifice plate and the top plate to insure airtightness.

Tupperwave Power-Take-Off

The Tupperwave PTO is composed of non-return valves, 2 accumulators and a unidirectional impulse turbine. Each reservoir is connected to the OWC chamber through a system of non-return valves and connected to one another by the turbine.

Scaling method of HP and LP chambers Since the device working principal relies on air compressibility in the HP and LP chambers, the right scaling method needs to be used for the down-scaling of the chambers. The scale ratio ε is 0.0415. A scaling method suggested to scale down the OWC chamber volume to properly represent the spring-like effect of the air in the OWC chamber is suggested in [8]. Assuming the turbines to be equally efficient at model and full scale, the ratio between the OWC chamber volume at model scale and full scale is given by:

$$\frac{V_{0M}}{V_{0F}} = \varepsilon^2 \cdot \delta^{-1} \quad (2)$$

δ is defined as the ratio between the water densities ρ_w at full and small scale ($\delta = \frac{\rho_{wM}}{\rho_{wF}} = 0.97$). Using the same derivation method as used in [8] it can be shown that this same method is applicable for the scaling of the accumulators chambers.

This scaling law for the chambers size requires much bigger size chamber than the Froude similarity would require. In the full scale device, the HP and LP chambers are 950 m^3 each. Froude scaling would give 0.068 m^3 for each chamber at small scale. The scaling suggested by equations 2 gives 1.69 m^3 per chamber. Unlike for the full scale, it is impossible to fit both chambers on the device as their volume largely exceed the overall volume of the device. The alternative at small scale is to locate the main volume of the HP and LP chambers outside of the device and connect them to two smaller chambers on the device with flexible pipes.

Two 1 m^3 IBC tanks per chamber were used and were partially filled with water in order to match the required volume of air per accumulator. Figure 3 shows the Tupperwave model in regular waves connected to both HP and LP accumulators on the pedestrian bridge. Care was taken to reduce the influence of the flexible pipes on the motion of the buoy and part of the pipes weight was supported by bungee ropes.

Remark on OWC chamber scaling method Since the air compressibility in the HP and LP chamber is essential in the Tupperwave device working-principle, efforts were made to physically model it by using the scaling method described above. However, the air compressibility in the full scale 148 m^3 OWC chamber, is not essential for the device working-principle and was therefore not modelled. This would have required a third flexible pipe connecting the OWC chamber to another 0.264 m^3 reservoir on the bridge and would have increase the testing difficulty even more. The OWC chamber was therefore downscaled with Froude similarity to 0.011 m^3 . Thus, this needs to be kept in mind that dynamic similarity of the air spring-like effect in the OWC chamber of the full scale device was not achieved in this testing. The study of the exact consequences of such modelling on the device behaviour is of interest for a future work.

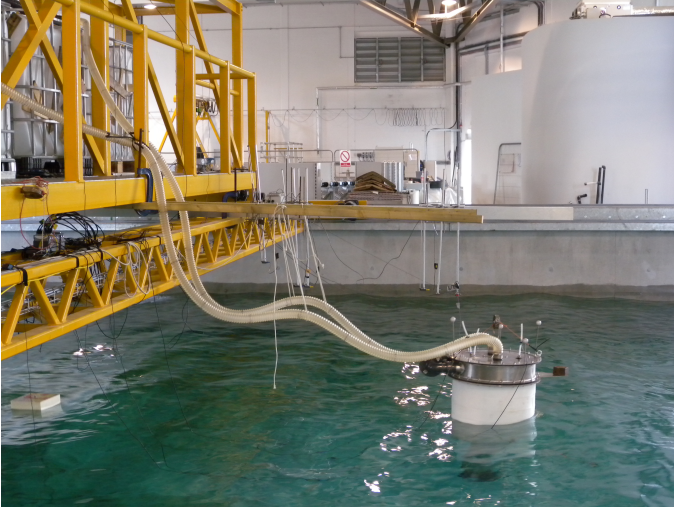


FIGURE 3. TUPPERWAVE IN IRREGULAR WAVES

Turbine The unidirectional turbine chosen for the Tupperwave device is also of impulse type and modelled using orifice plates located inbetween the two smaller chambers on the device. The numerical modelling carried out previously in [7] showed that the optimized turbine damping coefficient was in the range of $300 - 750 \text{ Pa.s}^2.\text{kg}^{-2}$ for the full scale device. For the model scale device 3 orifice plates were built around those values scaled down using Froude similarity law. Their exact damping coefficient were also determined experimentally prior to testing.

Valves The Tupperwave's working principle relies on the use of non-return valves. The valves are a key component of the PTO because they are likely to cause pneumatic power losses. They can either be passive or active. Passive valves are usually less complex to implement and more robust than active valves but may not be as efficient.

For the physical model, passive valves were chosen for their simplicity. The most appropriate valves found on the market for the small scale physical model were the Capricorn MiniHab HypAir-Balance, see fig. 4. They are passive normally closed air admittance valves from the plumbing market. A rubber membrane contained in the valve obstructs with gravity the opening of the valve. When sufficient pressure is applied, the rubber membrane is lifted up and the valve opens. Their opening pressure is about 70Pa (1686Pa at full scale) and their relatively small size allowed their use in the small scale physical model.

Their only drawback is the fact that, since they are gravity operated, they only work properly when positioned the right way up in vertical position. The HP valves therefore had to be positionned on a U-shaped PVC duct. Since a larger valve opening area reduces the valves damping, it is beneficial to maximise the number of valves on each side. Because of the necessary

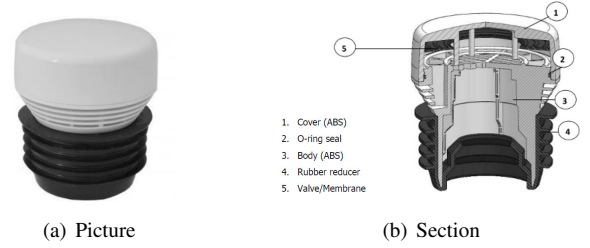


FIGURE 4. MINIHAB HYPAIRBALANCE FROM CAPRICORN USED IN TUPPERWAVE SMALL SCALE MODEL

TABLE 1. DEVICES MASS PROPERTIES

	Model scale	Full scale
Total mass (kg)	58.4	817.10^3
COG (m)	0.892	21.49
COB (m)	0.961	23.16
Ixx (kg.m^2)	23	$1,87.10^8$
Iyy (kg.m^2)	23.5	$1,91.10^8$
Izz (kg.m^2)	2	$1.62.10^7$

U-shaped PVC set-up for the HP valves, only 2 valves could be fitted between the OWC chamber and the HP chamber. In order to keep the device symmetrical, 2 valves were also fitted on the other side, between the LP chamber to the OWC chamber.

The damping of the valves will be discussed later in the article.

Devices mass properties The conventional OWC PTO being lighter than the Tupperwave PTO, the conventional OWC was ballasted such that both device have the exact same mass properties. The mass properties of the device are given in table 1.

EXPERIMENTAL SETUP

The device was tested in the LIR-NOTFs Deep Ocean Basin in fixed and floating configuration under regular and irregular sea states. The fixed configuration was only tested in the scope of the numerical model validation and is not described in this article.

Wave Basin

The dimensions of the LIR-NOTFs Deep Ocean Basin are 35m long, 12m wide and 3m deep. It has a movable floor plate to allow the water depth be adjusted, making it suitable for circa. 1/15 scale operational conditions and 1/50 scale survival waves

TABLE 2. TESTED BRETSCHNEIDER SEA STATE

Bretschneider	Hs (m)		Tp (s)	
	Full scale	Model	Full scale	Model
B1	2	0.083	5.66	0.083
B2	3	0.1245	7.07	0.1245
B3	3	0.1245	8.49	0.1245
B4	5	0.2075	8.49	0.2075
B5	3	0.1245	10.61	0.1245
B6	5	0.2075	10.61	0.2075
B7	5	0.2075	12.73	0.2075
B8	3	0.1245	14.14	0.1245

in Atlantic conditions. Equipped with 16 hinged force feedback paddles capable of a peak wave generation condition of $H_s = 0.6\text{m}$, $T_p = 2.7\text{s}$ and $H_{\max} = 1.1\text{m}$. An instrument bridge runs across the width of the basin and is used to support the electrical instrumentation and cables. A pedestrian bridge spans the basin as well and was used to support the four IBC tanks. For the tests the water depth was set to 2.075m , equivalent to 50m at full scale.

Test Plan

The devices were tested in both regular and irregular sea states. For the regular sea states, 2 wave heights (2 and 4m at full scale equivalent) were tested with periods ranging from 5 to 14s . A set of irregular sea states of various significant wave heights and peak periods was also tested. Table 2 displays the characteristics of the irregular wave tests.

The regular wave tests were 125 seconds long, which is equivalent to 10 minutes at full scale. The irregular wave tests are 7 minutes long which is equivalent to 35 minutes at full scale. This allows a full representation of the Bretschneider sea state.

For the analysis of regular wave tests, averaging of the key variables is made over several waves once a steady state is reached, practically between 90 and 115s at model scale. The average values in irregular sea states are calculated over the full time of the simulation.

Moorings

The device is moored with a 3 point mooring arrangement of catenary type with 120 degrees between any two mooring lines. There are 2 bow mooring lines and 1 stern line.

Device Monitoring

Both devices were equipped with a number of instrument to monitor the devices' behaviours. They were connected to a National Instrument Compact Rio data acquisition system which recorded at a sampling rate of 32Hz and stored the data on a text file. Pressure sensors, wave probes and 3D-cameras allowed to fully monitor the movements, pressures and flows of the device. For the measurement of the internal water surface relatively to the buoy (IWS), a wave probe was located in the water column. The air flows through the orifices and valves were calculated using the measured pressures. The mass flow \dot{m}_t across the thin orifice plate representing the turbine was calculated using eq. 1 by:

$$\dot{m}_t = \sqrt{\frac{\Delta p_t}{k_t}} \quad (3)$$

k_t is called orifice damping coefficient.

The same mathematical model was used for the valves:

$$\dot{m}_v = \begin{cases} \sqrt{\frac{\Delta p_v - p_{thres}}{k_v}} & \text{if } p_v > p_{thres} \\ 0 & \text{if } p_v < p_{thres} \end{cases} \quad (4)$$

k_v is called valve damping coefficient and p_{thres} is the valve opening pressure (70Pa).

The pneumatic power flowing through the orifice PTO P_t and through the valves P_v is calculated as the product of the volumetric flow and the pressure drop:

$$P = Q \cdot \Delta p = \frac{\dot{m}}{\rho_0} \cdot \Delta p \quad (5)$$

The available hydraulic power P_{hydro} is the power of the force applied by the IWS on the air contained in the OWC chamber:

$$P_{hydro} = S \cdot \frac{dx_{IWS}}{dt} \cdot p_{owc} \quad (6)$$

where S is IWS area, x_{IWS} is the position of the IWS relatively to the buoy and p_{owc} is the excess pressure of the air in the OWC chamber.

The available hydraulic power P_{hydro} is the hydraulic power extracted from the waves by the device and converted into available pneumatic power. P_v is the pneumatic power dissipated in the valves. P_t is the useful pneumatic power or pneumatic power available to the turbine.

TABLE 3. DEVICE NATURAL PERIODS OF OSCILLATIONS

Natural oscillation period (s)	Small scale	Full scale
Heave	1.37	6.7
Pitch	4.48	22.0
Roll	4.43	21.7

RESULTS AND ANALYSIS

In this section, the Tupperwave device behaviour is first described and the power losses in the valves are estimated. The performance of the Tupperwave device are then compared to the conventional OWC tested in parallel. All results are given at full scale equivalent to give the reader a more significant perspective of the devices performances.

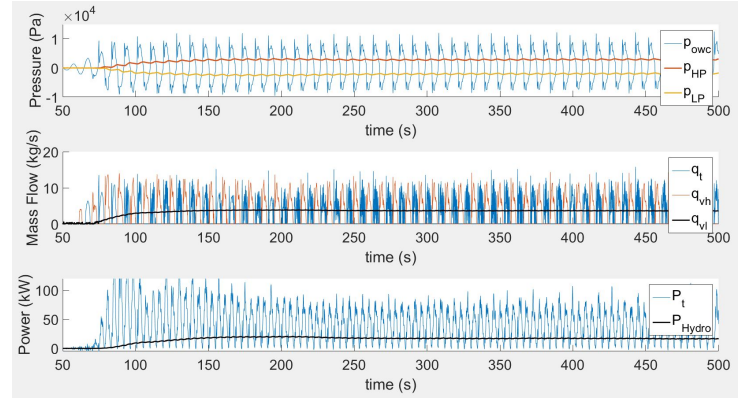
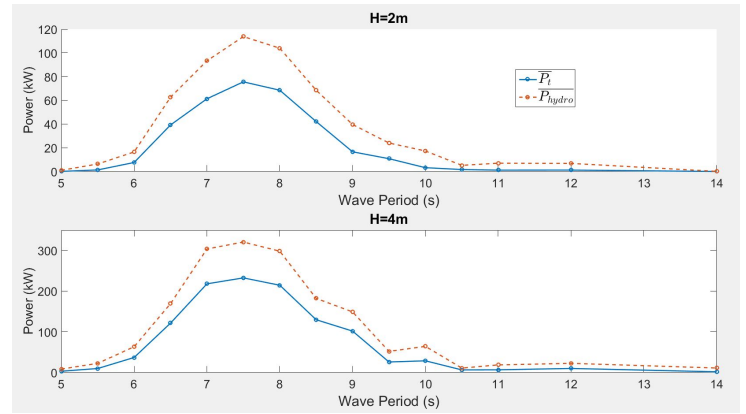
Tupperwave Device Behaviour

Decay tests Decay tests for the heave, pitch and roll motion of the device were performed. The PTO was not installed on top of the water column in order to suppress all water column damping which can influence the buoy movement. The natural periods of oscillation in these three degrees of freedom are given in table 3. The slight difference in the pitch and roll natural oscillation periods can be explained by the difference in moment of inertia due to the location of the Tupperwave orifice and its counter-balancing weight.

Regular Waves The Tupperwave device creates in regular waves a constant pressure difference between the HP and LP chamber which generates a smooth unidirectional flow across the orifice PTO. Among the 3 orifices tested, the device showed the best performance when equipped with a 9.2mm diameter of damping coefficient $1.24 \cdot 10^8 \text{ Pa} \cdot \text{s}^2 \cdot \text{kg}^{-2}$ which is equivalent to $367 \text{ Pa} \cdot \text{s}^2 \cdot \text{kg}^{-2}$ at full scale. The graphs of the pressure, mass flow and power time series obtained with this orifice for 2m high regular waves with 9 seconds period are given in fig. 5. p_{owc} , p_{HP} and p_{LP} are the excess pressures relatively to the atmosphere in the OWC chamber, the HP chamber and the LP chamber respectively. q_t , q_{vh} , and q_{vl} are the mass flows across the orifice and the HP and LP valves respectively (see fig. 1).

The power time series show that the Tupperwave PTO system transforms the highly fluctuating available hydraulic power into a smooth pneumatic power across the orifice. But the two signals do not have the same average value. It is clear that there has been losses in the transformation process.

Figure 6 displays the average available hydraulic power and the average pneumatic power available to the turbine in 2 and 4 meter high waves once the steady state was reached. The device produces the most useful pneumatic power for wave periods

**FIGURE 5. TUPPERWAVE DEVICE - 2 METER HIGH AND 9 SECONDS PERIOD REGULAR WAVES - PRESSURE, MASS FLOW AND POWER TIME SERIES****FIGURE 6. TUPPERWAVE DEVICE - AVERAGE AVAILABLE HYDRAULIC POWER AND USEFUL PNEUMATIC POWER**

around 6.5-8.5s. The device does not convert all hydraulic power into useful pneumatic power.

Figure 7 compares the average available hydraulic power to the sum of the average dissipated pneumatic power in the valves ($\bar{P}_{v_{hp}}$ and $\bar{P}_{v_{lp}}$) and in the orifice (\bar{P}_t), and shows that the available hydraulic power is entirely dissipated. This shows that the difference between the hydraulic power and the useful pneumatic power observed in fig. 6 is equal to the power dissipated in the HP and LP valves.

The valves efficiency is defined as the ratio between the average pneumatic power available to the turbine over the average available hydraulic power of the water column:

$$\eta_{valves} = \frac{\bar{P}_t}{\bar{P}_{hydro}} \quad (7)$$

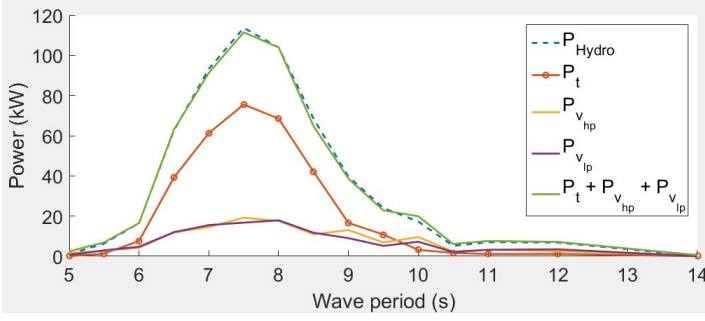


FIGURE 7. TUPPERWAVE DEVICE - AVERAGE AVAILABLE HYDRAULIC POWER AND PNEUMATIC DISSIPATED POWERS

Figure 7 has shown that:

$$\bar{P}_{hydro} = \bar{P}_t + \bar{P}_{v_{hp}} + \bar{P}_{v_{lp}} \quad (8)$$

Assuming that the valves have the same damping $k_v = k_{v_{hp}} = k_{v_{lp}}$, it can be shown that the valves efficiency can be approximated from the turbine and valve damping characteristics using the equation:

$$\eta_{valves} \simeq \frac{1}{1 + 23 \cdot \frac{k_v}{k_t} + 2 \cdot \frac{P_{thres}}{\Delta p_t}} \quad (9)$$

Figure 8 compares the actual efficiency to the estimation obtained with Eqn. 9 for $H=2m$ and $H=4m$. Equation 9 gives a good approximation of the actual valves efficiency. This formula shows that in order to maximise the Tupperwave valves efficiency, the opening pressure of the valves need to be small compared to the average pressure drop across the orifice and the damping coefficient of the valves needs to be small compared to the damping coefficient of the turbine.

The valves efficiency is maximised between 6.5-8.5 seconds where the spar buoy and water column excitation is maximum. Outside of this range, the efficiency quickly drops and the hydraulic power from the water column is not transferred to the air in the OWC chamber. The device is also more efficient in higher waves. A maximum of 72.5% efficiency is reached for $H=4m$. This is due to the passive valves that require sufficient pressure to open fully.

Higher valve efficiency (up to 80%) was obtained for the orifice with higher damping coefficient k_t which created higher average pressure drop Δp_t across the turbine. These features clearly increase the efficiency according to equation 9. But this orifice allowed less hydraulic power to be extracted from the waves and produced in the end less pneumatic power. Figure 9 displays the average damping coefficient k_v of the HP valves obtained

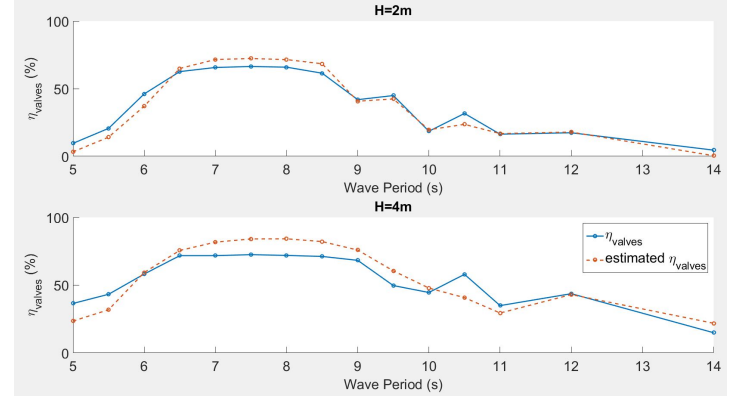


FIGURE 8. TUPPERWAVE DEVICE - REGULAR WAVES - VALVES EFFICIENCY

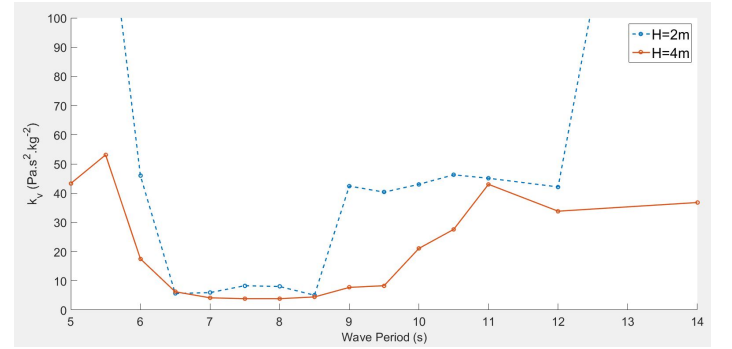


FIGURE 9. TUPPERWAVE DEVICE - REGULAR WAVES - HP VALVES DAMPING COEFFICIENT

for the different wave periods. The damping coefficient of the valves reaches a plateau around $5 \text{ Pa.s}^2.\text{kg}^{-2}$ between 6.5 and 8.5s wave period. It is larger outside of this range and thus the valves efficiency decreases. For $H=4m$, the valves open fully on a larger range of wave periods. The dependence of the valves efficiency on the bodies excitation is clearly a drawback to passive valves because it reduces the device performance bandwidth.

The Tupperwave model creates a steady pressure head across the orifice and converts the highly fluctuating hydraulic power extracted from the waves into a smooth pneumatic power across the orifice. A maximum of 15kW of pneumatic power per meter square wave height is obtained for wave periods between 7 and 8s at full scale equivalent. The damping coefficient of the passive valves varies with the bodies excitations and significant pneumatic power dissipation happens in the valves. For the best performing orifice, a maximum of 72.5% of the hydraulic power extracted from the waves is actually converted into useful pneumatic power. A formula to estimate the valves efficiency from the valves characteristic was derived and showed the importance

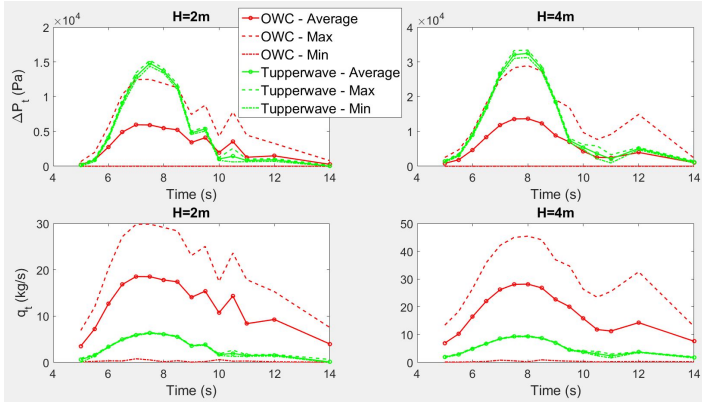


FIGURE 10. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - REGULAR WAVES - PRESSURE HEAD AND MASS FLOW ACROSS THE ORIFICE

of the opening pressure and damping coefficient in the valves efficiency. The valves used in these tests are plumbing valves that have not been designed for such application. It is therefore believed that it is possible to improve the Tupperwave device performances by improving the valves and reduce significantly the amount of losses.

Tupperwave Device and Conventional OWC Power Performances

In this section, the performances of the floating conventional OWC and floating Tupperwave are compared. The devices were built using the same Spar OWC structure in order to provide a fair comparison in terms of power performance. Both devices are compared with their respective orifice that maximised their pneumatic power output.

Regular waves The conventional OWC and Tupperwave device behave differently in converting the wave energy into useful pneumatic energy. Average pressure head and air flow rate across the turbines are compared in fig. 10. The maximum and minimum values of the pressures and flow rates in steady state are also displayed such that to understand their fluctuations. The conventional OWC produces large flows and small pressure drops compared to Tupperwave device. The pressure drops and flows produced by the conventional OWC are largely fluctuating from 0 to twice their average value while the fluctuations from the Tupperwave device are barely visible. The turbine requirements of both devices are therefore very different. The Tupperwave turbine is likely to be smaller because of the reduced flow.

Figure 11 compares available hydraulic power and pneumatic power available to the turbine for the conventional OWC and the Tupperwave model at H=2m and 4m. The Tupperwave model extracts less hydraulic power from the waves than the

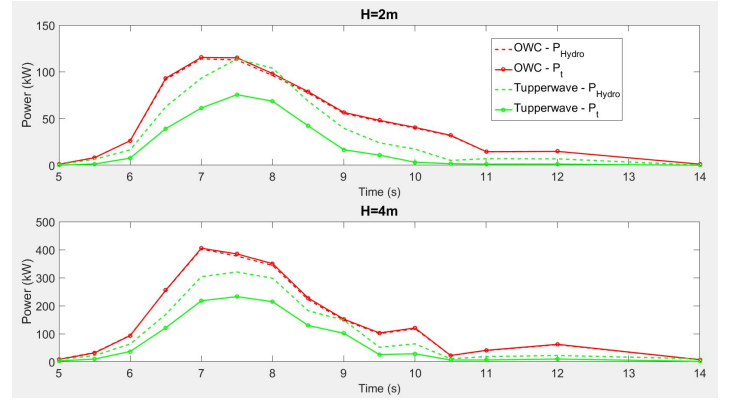


FIGURE 11. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - REGULAR WAVES - HYDRAULIC AND PNEUMATIC POWER OUTPUT

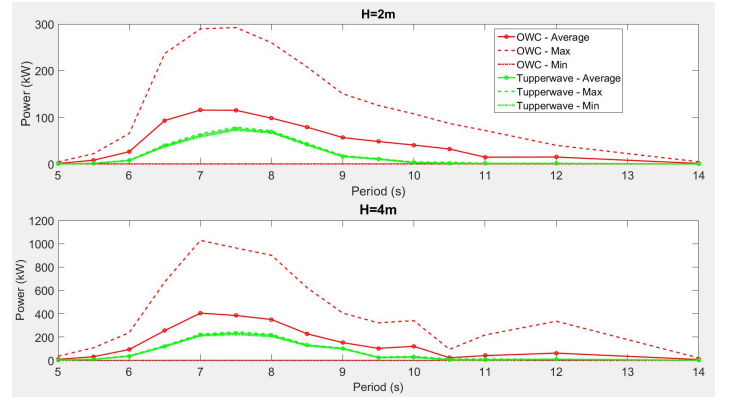


FIGURE 12. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - REGULAR WAVES - PNEUMATIC POWER OUTPUT

conventional OWC. The conventional OWC converts entirely the available hydraulic power into useful pneumatic power. The Tupperwave model converts only 60-70% of the hydraulic power into useful pneumatic power. The Tupperwave model produces in the end between 30 and 70% of the useful pneumatic power generated by the conventional OWC for wave periods from 6 to 9 seconds. The useful pneumatic power produced by the Tupperwave device is however much smoother than for the conventional OWC. Figure 12 displays the average, maximum and minimum values of the useful pneumatic power in steady state. The standard deviation of the pneumatic power around its average value is of 2% for the Tupperwave device against 87% for the conventional OWC.

Various types of self-rectifying turbines have been developed for the pneumatic to electrical power conversion in conventional OWCs. The most recent and most efficient prototypes are the biradial and twin-rotor turbine. Their efficiencies

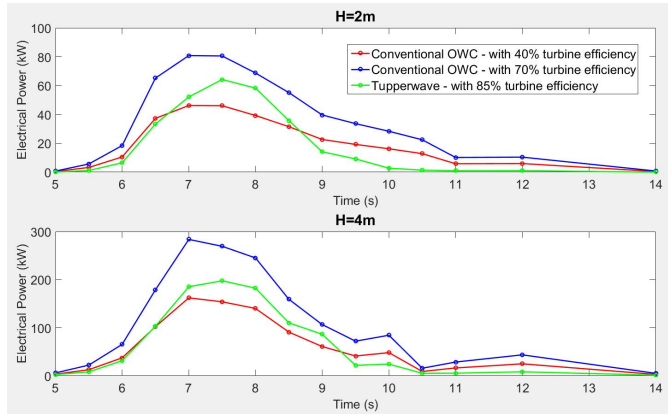


FIGURE 13. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - REGULAR WAVES - ELECTRICAL POWER OUTPUT ASSUMING TURBINES

in constant air flows was assessed experimentally to be 75% and 80% respectively. In real flow conditions, their efficiency is lower and the maximum efficiency of the biradial turbine in irregular flow conditions was estimated of 70% [9, 10]. Older versions of fixed-guide vanes self-rectifying impulse turbines have reached 40% efficiency in real flow conditions [11, 12].

The Tupperwave device uses a single-stage unidirectional turbines, such turbines reach 85% efficiency in constant flow conditions [10, 13]. The flow in the Tupperwave device is relatively constant as can be seen in the previous section. Hence, the turbine is likely to operate at maximum efficiency most of the time.

From the useful pneumatic power obtained in tank testing, it is possible to approximate the electrical power that would be created if the devices were equipped with a turbine. This is done by multiplying the useful pneumatic power created in the tank testing to the assumed turbines efficiencies with the assumption that the generators are 100% efficient. The comparison between the theoretical electrical power produced in regular sea states by both devices is shown in fig. 13. This method showed that the Tupperwave device produces less electrical power than the conventional OWC equipped with the most efficient self-rectifying turbine. However, the valves used in the testing have not been designed for this purpose and are causing 20 to 50% pneumatic power losses depending on the device excitation. A proper design of appropriate valves would enhance the electrical power production of the Tupperwave device.

Irregular waves The comparison of power performances between conventional OWC and Tupperwave in irregular sea states is shown in fig. 14. The results of the Tupperwave device compared to the conventional OWC are not as good as for

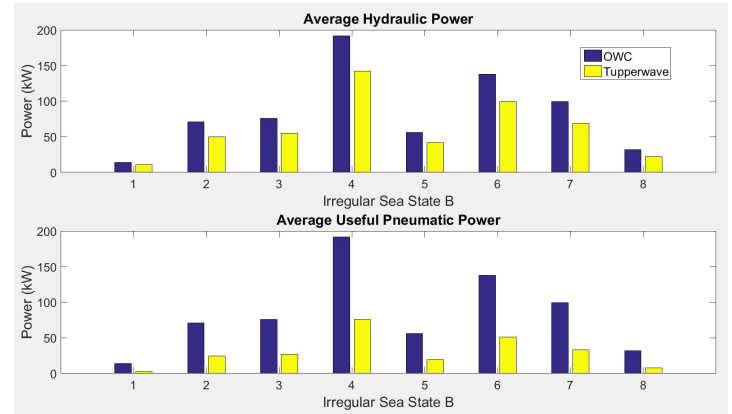


FIGURE 14. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - IRREGULAR WAVES - HYDRAULIC AND PNEUMATIC POWER OUTPUT

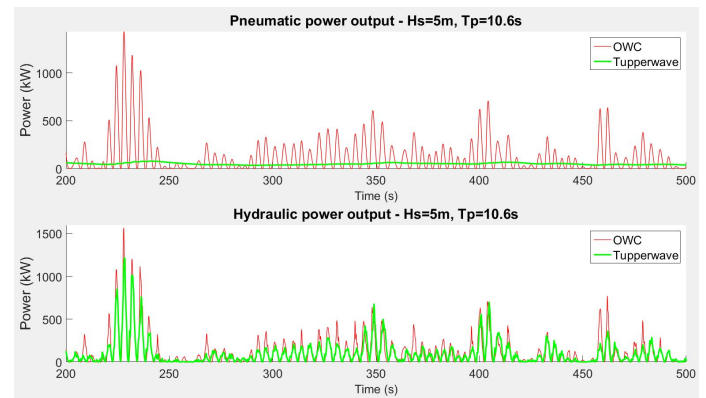


FIGURE 15. TUPPERWAVE DEVICE AND CONVENTIONAL OWC - IRREGULAR WAVES - HYDRAULIC AND PNEUMATIC POWER TIME SERIES

the regular waves. The Tupperwave device produces in the end only about 35% of the useful pneumatic power produced by the conventional OWC. The efficiency of the Tupperwave valves for the irregular sea states shown to be about 50%. This is due to the fact that the part of the energy contained in the irregular wave series is transported by the smaller waves for which the valves do not open properly or do not open at all. The valves used in these tests are the cause of the poor performance of the Tupperwave device in irregular sea states. The smoothing of the pneumatic power is however still remarkable as shown in fig. 15.

Conclusion

A 1/24th scale Tupperwave device was built and tested in the Lir-NOTF's Deep Ocean Basin. The scaling method used to correctly simulate the air compressibility in the high and low

pressure chambers involved big reservoirs and flexible pipes to connect the device to the reservoirs. A conventional Spar buoy OWC was also built on the same geometry as the Tupperwave model in order to provide a fair comparison between the two devices pneumatic power performances. Both devices were tested in regular and irregular sea states. Their behaviour and pneumatic power outputs were monitored and compared.

In regular waves, the smoothing of the pneumatic power by the Tupperwave PTO is significant: 2% pneumatic power fluctuation around the mean value across the turbine is obtained against 87% for the conventional OWC. The Tupperwave device produced however less useful pneumatic power. It produced in average 30 to 70% of the useful pneumatic power produced by the conventional OWC. The main losses are the pneumatic power dissipation through the valves. The efficiency of the valves used in these tests to convert water column hydraulic energy into useful pneumatic energy was in the range of 50% to 80% depending on the orifice and on the device excitation.

In irregular sea states the Tupperwave device produced about 35% of the useful pneumatic energy produced by the conventional OWC. The main reason for that is the poor efficiency of the valves which only worked properly for high-energetic wave groups and did not allow energy extraction from the waves for the rest of the spectrum.

The valves used in the Tupperwave model were basic air admittance valves found on the plumbing market. These valves were not designed specifically for this purpose and there is therefore large room for improvement. Nevertheless, the Tupperwave device produced decent pneumatic power in regular waves with remarkable smoothness which would allow the unidirectional turbine to be fully efficient with relatively simple control law. These tests revealed the importance of the valves for the Tupperwave power performance. A formula to estimate the passive valves efficiency from the orifice and valves characteristics was suggested. Low damping and quick opening is key to maximise the power production. Well designed valves are the conditions for Tupperwave device to be competitive against conventional OWCs equipped with high-end self-rectifying turbines in terms of power production.

Future work will focus on the validation of the numerical model developed in [7] using the tank testing results described in this article. A Wave-to-Wire numerical model will then be developed to accurately compare the electrical power performance of the Tupperwave device to the conventional OWC.

ACKNOWLEDGMENT

The authors would like to acknowledge funding received through OCEANERA-NET European Network (OCN/00028).

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